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$$\Delta p_{fan\ eff} = \Delta p_{fan} + [0.8-(-0.43)]\frac{0.075(23.2)^2}{2(32.2)}$$

$$= \Delta p_{fan} + 0.77\ \mathrm{lb_f/ft^2}$$

This wind-assisted hourly averaged pressure is exceeded only 1% of the time (88 hours per year). When wind direction reverses, the outlet will be on the upwind wall and the inlet on the downwind wall, producing wind-opposed flow, changing the sign from −0.15 to −0.15 in. of water. The importance of these pressures depends on their size relative to the fan pressure rise Δp_{fan}, as shown in Figure 13.

Minimizing Wind Effect on System Volume

Wind effect can be reduced by careful selection of inlet and exhaust locations. Because wall surfaces are subject to a wide variety of positive and negative pressures, wall openings should be avoided when possible. When they are required, wall openings should be away from corners formed by building wings (see Figure 11). Mechanical ventilation systems should operate at a pressure high enough to minimize wind effect. Low-pressure systems and propeller exhaust fans should not be used with wall openings unless their ventilation rates are small or they are used in noncritical services (e.g., storage areas).

Although roof air intakes in flow recirculation zones best minimize wind effect on system flow rates, current and future air quality in these zones must be considered. These locations should be avoided if a contamination source exists or may be added in the future. The best area is near the middle of the roof, because the negative pressure there is small and least affected by changes in wind direction (see Figure 8). Avoid edges of the roof and walls, where large pressure fluctuations occur. Either vertical or horizontal (mushroom) openings can be used. On roofs with large areas, where intake may be outside the roof recirculation zone, mushroom or 180° gooseneck designs minimize impact pressure from wind flow. Vertical louvered openings or 135° goosenecks are undesirable for this purpose or for rain protection.

Heated air or contaminants should be exhausted vertically through stacks, above the roof recirculation zone. Horizontal, louvered (45° down), and 135° gooseneck discharges are undesirable, even for heat removal systems, because of their sensitivity to wind effects. A 180° gooseneck for hot-air systems may be undesirable because of air impingement on tar and felt roofs. Vertically discharging stacks in a recirculation region (except near a wall) have the advantage of being subjected only to negative pressure created by wind flow over the tip of the stack. See Chapter 44 of the 2003 *ASHRAE Handbook—HVAC Applications* for information on stack design.

Chemical Hood Operation

Wind effects can interfere with safe chemical hood operation. Supply volume variations can cause both disturbances at hood faces and a lack of adequate hood makeup air. Volume surges, caused by fluctuating wind pressures acting on the exhaust system, can cause momentary inadequate hood exhaust. If highly toxic contaminants are involved, surging is unacceptable. The system should be designed to eliminate this condition. On low-pressure exhaust systems, it is impossible to test the hoods under wind-induced, surging conditions. These systems should be tested during calm conditions for safe flow into the hood faces, and rechecked by smoke tests during high wind conditions. For more information on chemical hoods, see Chapter 14 of the 2003 *ASHRAE Handbook—HVAC Applications*. For more information on stack and intake design, see Chapter 44 of that volume.

BUILDING PRESSURE BALANCE AND INTERNAL FLOW CONTROL

Proper building pressure balance avoids flow conditions that make doors hard to open and cause drafts. In some cases (e.g., office buildings), pressure balance may be used to prevent confinement of contaminants to specific areas. In other cases (e.g., laboratories), the correct internal airflow is towards the contaminated area.

Pressure Balance

Although supply and exhaust systems in an internal area may be in nominal balance, wind can upset this balance, not only because of its effects on fan capacity but also by superimposing infiltrated or exfiltrated air (or both) on the area. These effects can make it impossible to control environmental conditions. Where building balance and minimum infiltration are important, consider the following:

- Design HVAC system with pressure adequate to minimize wind effects.
- Include controls to regulate flow rate, pressure, or both.
- Separate supply and exhaust systems to serve each building area requiring control or balance.
- Use revolving or other self-closing doors or double-door air locks to noncontrolled adjacent areas, particularly outside doors.
- Seal windows and other leakage sources.
- Close natural ventilation openings.

Internal Flow Control

Airflow direction is maintained by controlling pressure differentials between spaces. In a laboratory building, for example, peripheral rooms such as offices and conference rooms are kept at positive pressure, and laboratories at negative pressure, both with reference to corridor pressure. Pressure differentials between spaces are normally obtained by balancing supply system airflows in the spaces in conjunction with exhaust systems in the laboratories. Differential pressure instrumentation is normally used to control the airflow.

The pressure differential for a room adjacent to a corridor can be controlled using the corridor pressure as the reference. Outdoor pressure cannot usually control pressure differentials within internal spaces, even during periods of relatively constant wind velocity (wind-induced pressure). A single pressure sensor can measure the outside pressure at one point only and may not be representative of pressures elsewhere.

Airflow (or pressure) in corridors is sometimes controlled by an outdoor reference probe that senses static pressure at doorways and air intakes. The differential pressure measured between the corridor and the outside may then signal a controller to increase or decrease airflow to (or pressure in) the corridor. Unfortunately, it is difficult to locate an external probe where it will sense the proper external static pressure. High wind velocity and resulting pressure changes around entrances can cause great variations in pressure.

To measure ambient static pressure, the probe should be located where airflow streamlines are not affected by the building or nearby buildings. One possibility is at a height of 1.5*R*, as shown in Figure 1. However, this is usually not feasible. If an internal space is to be pressurized relative to ambient conditions, the pressure must be known on each exterior surface in contact with the space. For example, a room at the northeast corner of the building should be pressurized with respect to pressure on both the north and east building faces, and possibly the roof. In some cases, multiple probes on a single building face may be required. Figures 4 to 8 may be used as guides in locating external pressure probes. System volume and pressure control is described in Chapter 46 of the 2003 *ASHRAE Handbook—HVAC Applications*.

NO FLOOR TO FLOOR COMPARISON

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Variable air volume with reheat permits airflow to be reduced as the first step in control; heat is then initiated as the second step. Compared to constant-volume reheat, this procedure reduces operating cost appreciably because the amount of primary air to be cooled and secondary air to be heated is reduced. Many types of controls can provide control sequences with more than one minimum airflow. This type of control allows the box to go to a lower flow rate that just meets ventilation requirements at the lightest cooling loads, then increase to a higher flow rate when the heating coil is energized, further reducing reheat energy use. A feature can be provided to isolate the availability of reheat during the summer, except in situations where even low airflow would overcool the space and should be avoided or where increased humidity causes discomfort (e.g., in conference rooms when the lights are turned off).

Because the reheat coil requires some minimum airflow to deliver heat to the space, and because the reheat coil must absorb all of the cooling capacity of that minimum airflow before it starts to deliver heat to the space, energy use can be significantly higher than with throttling boxes that go fully closed.

Induction. The VAV induction system uses a terminal unit to reduce cooling capacity by simultaneously reducing primary air and inducing room or ceiling air (replacing the reheat coil) to maintain a relatively constant room supply volume. This operation is the reverse of the bypass box. The primary-air quantity decreases with load, retaining the savings of VAV, and the air supplied to the space is kept relatively constant to avoid the effect of stagnant air or low air movement. VAV induction units require a higher inlet static pressure, which requires more fan energy, to achieve the velocities necessary for induction.

Fan-Powered. Fan-powered systems are available in either parallel or series airflow. In **parallel-flow** units, the fan is located outside the primary airstream to allow intermittent fan operation. A backdraft damper on the terminal fan prevents conditioned air from escaping into the return air plenum when the terminal fan is off. In **series** units, the fan is located in the primary airstream and runs continuously when the zone is occupied. These constant-airflow fan boxes in a common return plenum can help maintain indoor air quality by recirculating unvitiated air from overventilated zones to zones with greater outside air ventilation requirements.

Fan-powered systems, both series and parallel, are often selected because they maintain higher air circulation through a room at low loads but still retain the advantages of VAV systems. As the cold primary-air valve modulates from maximum to minimum (or closed), the unit recirculates more plenum air. In a perimeter zone, a hot-water heating coil, electric heater, baseboard heater, or remote radiant heater can be sequenced with the primary-air valve to offset external heat losses. Between heating and cooling operations, the fan only recirculates ceiling air. This permits heat from lights to be used for space heating for maximum energy saving. During unoccupied periods, the main supply air-handling unit remains off and individual fan-powered heating zone terminals are cycled to maintain required space temperature, thereby reducing operating cost during unoccupied hours.

Fans for fan-powered air-handling units operated in series are sized and operated to maintain minimum static pressures at the unit inlet connections. This reduces the fan energy for the central air handler, but the small fans in fan-powered units are less efficient than the large air handler fans. As a result, the series fan-powered unit (where small fans operate continuously) may use more fan energy than a throttling unit system. However, the extra fan energy may be more than offset by the reduction in reheat through the recovery of plenum heat and the ability to operate a small fan to deliver heat during unoccupied hours where heat is needed.

Because fan-powered boxes involve an operating fan, they may generate higher sound levels than throttling boxes. Acoustical ceilings generally are not very effective sound barriers, so extra care should be taken in considering the sound level in critical spaces near fan-powered terminal units.

Both parallel and series fan-powered terminal units may be provided with filters. The constant (series) fan VAV terminal can accommodate minimum (down to zero) flow at the primary-air inlet while maintaining constant airflow to the space.

Both types of fan-powered units and induction terminal units are usually located in the ceiling plenum to recover heat from lights. This allows these terminals to be used without reheat coils in internal spaces. Perimeter zone units are sometimes located above the ceiling of an interior zone where heat from the lights maintains a higher plenum temperature. Provisions must still be made for morning warm-up and night heating. Also, interior spaces with a roof load must have heat supplied either separately in the ceiling or at the terminal.

Terminal Humidifiers

Most projects requiring humidification use steam. This can be centrally generated as part of the heating plant, where potential contamination from water treatment of the steam is more easily handled and therefore of less concern. Where there is a concern, local generators (e.g., electric or gas) that use treated water are used. Compressed-air and water humidifiers are used to some extent, and supersaturated systems are used exclusively for special circumstances, such as industrial processes. Spray-type washers and wetted coils are also more common in industrial facilities. When using water directly, particularly in recirculating systems, the water must be treated to avoid dust accumulation during evaporation and the build-up of bacterial contamination.

Terminal Filters

In addition to air-handling unit filters, terminal filters may used at the supply outlets to protect particular conditioned spaces where an extra-clean environment is desired. Chapter 24 discusses this topic in detail.

AIR DISTRIBUTION SYSTEM CONTROLS

Controls should be automatic and simple for best operating and maintenance efficiency. Operations should follow a natural sequence. Depending on the space need, one controlling thermostat closes a normally open heating valve, opens the outside air mixing dampers, or opens the cooling valve. In certain applications, an enthalpy controller, which compares the heat content of outside air to that of return air, may override the temperature controller. This control opens the outside air damper when conditions reduce the refrigeration load. On smaller systems, a dry-bulb control saves the cost of the enthalpy control and approaches these savings when an optimum changeover temperature, above the design dew point, is established. Controls are discussed in more detail in Chapter 46 of the 2003 *ASHRAE Handbook—HVAC Applications*.

NO FLOOR TO FLOOR COMPARISON

Air-handling systems, especially variable air volume systems, should include means to measure and control the amount of outside air being brought in to ensure adequate ventilation for acceptable indoor air quality. Strategies include the following:

- Separate constant-volume 100% outside air ventilation systems
- Outside air injection fan
- Directly measuring the outside air flow rate
- Modulating the return damper to maintain a constant pressure drop across a fixed outside air orifice
- Airflow-measuring systems that measure both supply and return air volumes and maintain a constant difference between them.
- CO_2- and/or VOC-based demand-controlled ventilation

A minimum outside air damper with separate motor, selected for a velocity of 1500 fpm, is preferred to one large outside air damper with minimum stops. A separate damper simplifies air balancing. Proper selection of outside, relief, and return air dampers is critical

for efficient operation. Most dampers are grossly oversized and are, in effect, unable to control. One way to solve this problem is to provide maximum and minimum dampers. A high velocity across a wide-open damper is essential to its providing effective control.

A mixed-air temperature control can reduce operating costs and also reduce temperature swings from load variations in the conditioned space. Chapter 46 of the 2003 *ASHRAE Handbook—HVAC Applications* shows control diagrams for various arrangements of central system equipment. Direct digital control (DDC) is common, and most manufacturers offer either a standard or optional DDC package for equipment, including air-handling units, terminal units, etc. These controls offer considerable flexibility. DDC controls offer the additional advantage of the ability to record actual energy consumption or other operating parameters of various components of the system, which can be useful for optimizing control strategies.

Constant-Volume Reheat. This system typically uses two subsystems for control: one controls the discharge air conditions from the air-handling unit, and the other maintains the space conditions by controlling the reheat coil.

Variable Air Volume. Air volume can be controlled by duct-mounted terminal units serving multiple air outlets in a control zone or by units integral to each supply air outlet.

Pressure-independent volume-regulator units control flow in response to the thermostat's call for heating or cooling. The required flow is maintained regardless of fluctuation of the VAV unit inlet or system pressure. These units can be field- or factory-adjusted for maximum and minimum (or shutoff) air settings. They operate at inlet static pressures as low as 0.2 in. of water.

Pressure-dependent devices control air volume in response to a unit thermostatic (or relative humidity) device, but flow varies with the inlet pressure variation. Generally, airflow oscillates when pressure varies. These units do not regulate flow but position the volume-regulating device in response to the thermostat. They are the least expensive units but should only be used where there is no need for maximum or minimum limit control and when the pressure is stable.

The type of controls available for VAV units varies with the terminal device. Most use either pneumatic or electric controls and may be either self-powered or system-air-actuated. Self-powered controls position the regulator by using liquid- or wax-filled power elements. System-powered devices use air from the air supplied to the space to power the operator. Components for both control and regulation are usually contained in the terminal device.

To conserve power and limit noise, especially in larger systems, fan operating characteristics and system static pressure should be controlled. Many methods are available, including fan speed control, variable-inlet vane control, fan bypass, fan discharge damper, and variable-pitch fan control. The location of pressure-sensing devices depends, to some extent, on the type of VAV terminal unit used. Where pressure-dependent units without controllers are used, the system pressure sensor should be near the static pressure midpoint of the duct run to ensure minimum pressure variation in the system. Where pressure-independent units are installed, pressure controllers may be at the end of the duct run with the highest static pressure loss. This sensing point ensures maximum fan power savings while maintaining the minimum required pressure at the last terminal.

As flow through the various parts of a large system varies, so does static pressure. Some field adjustment is usually required to find the best location for the pressure sensor. In many systems, the initial location is two-thirds to three-fourths of the distance from the supply fan to the end of the main trunk duct. As the pressure at the system control point increases as terminal units close, the pressure controller signals the fan controller to position the fan volume control, which reduces flow and maintains constant pressure. Many systems measure flow rather than pressure and, with the development of economical DDC, each terminal unit (if necessary) can be monitored and the supply and return air fans modulated to exactly match the demand.

Dual-Duct. Because dual-duct systems are generally more costly to install than single-duct systems, their use is less widespread. DDC, with its ability to maintain set points and flow accurately, can make dual-duct systems worthwhile for certain applications. They should be seriously considered as alternatives to single-duct systems.

Personnel. The skill levels of personnel operating and maintaining the air conditioning and controls should be considered. In large research and development or industrial complexes, experienced personnel are available for maintenance. On small and sometimes even large commercial installations, however, office managers are often responsible, so designs must be in accordance with their capabilities.

Water System Interface. On large hydronic installations where direct blending is used to maintain (or reset) the secondary-water temperature, the system valves and coils must be accurately sized for proper control. Many designers use variable flow for hydronic as well as air systems, so the design must be compatible with the air system to avoid operating problems.

Relief Fans. In many applications, relief or exhaust fans can be started in response to a signal from the economizer control or to a space pressure controller. The main supply fan must be able to handle the return air pressure drop when the relief fan is not running.

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2003 ASHRAE APPLICATIONS HANDBOOK (S)

CHAPTER 46

DESIGN AND APPLICATION OF CONTROLS

AUTOMATIC control of HVAC systems and equipment usually includes control of temperature, humidity, pressure, and flow rate. Automatic control sequences equipment operation to meet load requirements and provides safe operation of the equipment, using pneumatic, mechanical, electrical, electronic, and direct digital control devices.

This chapter covers (1) control of HVAC elements, (2) control of typical systems, (3) limit control for safe operation, and (4) design of controls for specific HVAC applications. Chapter 15 of the 2001 *ASHRAE Handbook—Fundamentals* covers the basics of control, types of control components, and commissioning of control systems.

CONTROL OF HVAC ELEMENTS

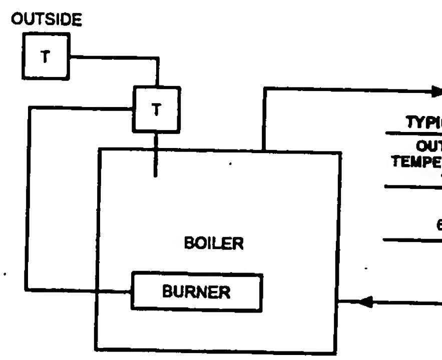
Boiler

Load affects the rate of heat input to a hydronic system. Rate control is accomplished by cycling and modulating the flame and by turning boilers on and off. Flame cycling and modulation are handled by the boiler control package. The control designer decides under what circumstances to add or drop a boiler and at what temperature to control the boiler supply water.

Hot-water distribution control includes temperature control at the hot-water boilers or converter, reset of heating water temperature, and control for multiple zones. Other factors that need to be considered include (1) minimum water flow through the boilers, (2) protection of boilers from temperature shock, and (3) coil freeze protection. If multiple or alternative heating sources (such as condenser heat recovery or solar storage) are used, the control strategy must also include a means of sequencing hot-water sources or selecting the most economical source.

Figure 1 shows a system for load control of a gas or oil-fired boiler. Boiler safety controls usually include flame-failure, high-temperature, and other cutouts. Intermittent burner firing usually controls capacity, although fuel input modulation is common in larger systems. In most cases, the boiler is controlled to maintain a constant water temperature, although an outside air thermostat can reset the temperature if the boiler is not used for domestic water heating. Figure 1 contains a typical reset schedule. To minimize condensation of flue gases and boiler damage, water temperature should not be reset below that recommended by the manufacturer, typically 140°F. Larger systems with sufficiently high pump operating costs can use variable-speed pump drives, pump discharge valves with minimum-flow bypass valves, or two-speed drives to reduce secondary pumping capacity to match the load.

Hot-water heat exchangers or steam-to-water converters are sometimes used instead of boilers as hot-water generators. Converters typically do not include a control package; therefore, the engineer must design the control scheme. The schematic in Figure 2 can be used with either low-pressure steam or boiler water ranging from 200 to 260°F. The supply water thermostat controls a modulating



TYPICAL RESET SCHEDULE	
OUTSIDE TEMPERATURE, °F	HOT-WATER SERVICE, °F
0	180
60	140

Fig. 1 Boiler Control

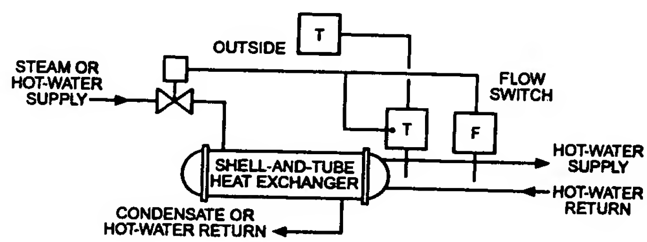


Fig. 2 Steam-to-Water Heat Exchanger Control

two-way valve in the steam (or hot-water) supply line. An outside thermostat usually resets the supply water temperature downward as the load decreases to improve the controllability of heating valves at low load and to reduce piping losses. A flow switch interlock should close the two-way valve when the hot-water pump is not operating. With integrated computer-based control, feedback from zone heating valves can be used to control the starting and stopping of the hot-water pumps. On constant-flow systems, the feedback can be used to reset the hot-water temperature to the lowest temperature that meets zone requirements.

Fan

The most efficient way to change the output of a fan is to change its speed. Because of their simplicity and high efficiency, variable-frequency drives are widely used. Though less efficient, eddy current drives are also an option for electronically controlling fan speed. Other ways of controlling fan ouput include using inlet cones, inlet guide vanes, and discharge or scroll dampers. Axial fans can be controlled by varying the pitch of the blade. Also, dampers and ducting can simply bypass some of the air from the supply side of the fan to the return side (Figure 3). Bypassing does not change the output of the fan, but it can allow the fan to accommodate flow variations in the distribution system without fan instability. The final selection of a control device is determined by efficiency requirements and available funding.

The preparation of this chapter is assigned to TC 1.4, Control Theory and Application.

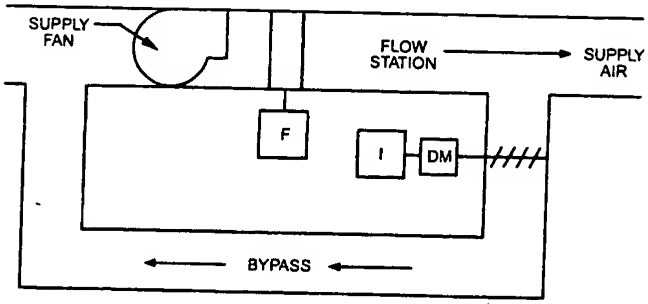


Fig. 3 Fan Bypass Control to Prevent Supply Fan Instability

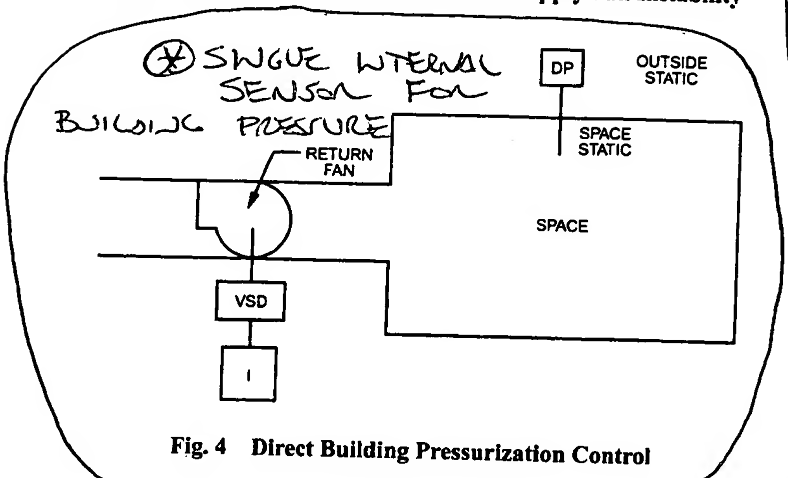


Fig. 4 Direct Building Pressurization Control

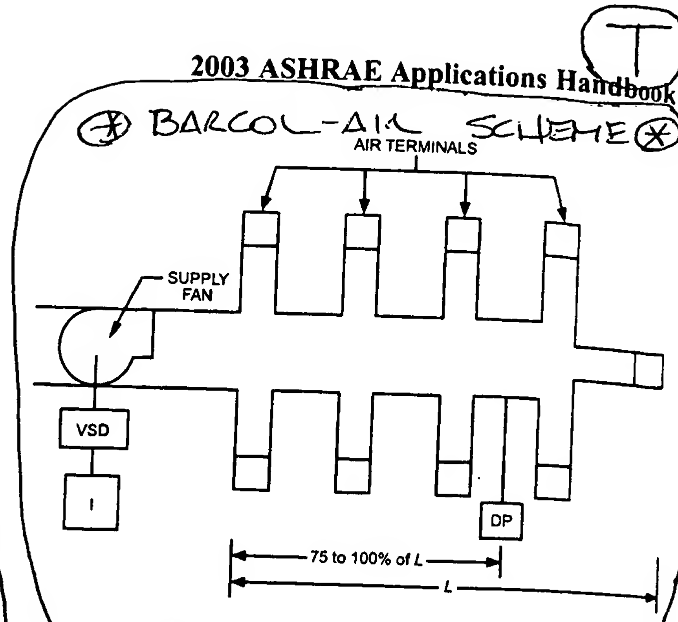


Fig. 5 Duct Static-Pressure Control

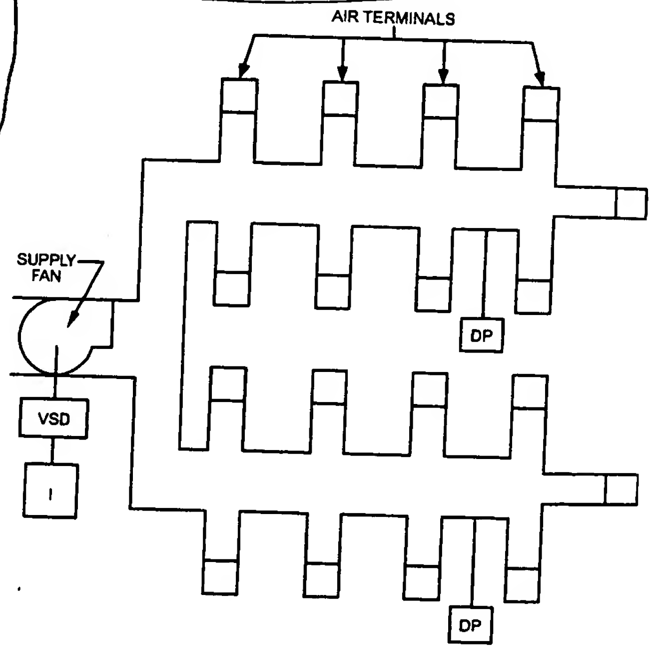


Fig. 6 Multiple Static Sensors

Static-pressure control is required in systems having variable flow rates. To conserve fan energy, the static-pressure controller should be set at the lowest control point permitting proper air distribution at design conditions. The controller requires proportional-plus-integral (PI) control because it eliminates offset while maintaining stability. In proportional-only control, the low proportional gain required to stabilize fan control loops allows static pressure to offset upward as the load decreases, which causes the supply fan to consume more energy.

BUILDING PRESSURE ONLY USES A SINGLE INTERNAL SENSOR

Differential static-pressure control is used to **pressurize a building** or space relative to adjacent spaces or the outside. Typical applications include clean rooms (positive pressure to prevent infiltration), laboratories (positive or negative, depending on use), and various manufacturing processes, such as spray-painting rooms. The pressure controller usually modulates dampers in the supply duct to maintain the desired pressure as exhaust volumes change. A method for control of the return fan requires measuring the space and outside static pressures (Figure 4). The location for measuring inside static pressure must be selected carefully: away from doors and openings to the outside, away from elevator lobbies, and, when using a sensor, in a large representative area shielded from drafts. The outside location must likewise be selected carefully, typically 10 to 15 ft above the building and oriented to minimize wind effects from all directions. The amount of minimum outside air varies with building permeability and exhaust fan operation. Control of building pressurization can affect the amount of outside air entering the building.

BARCOL AIR

Duct static-pressure control for variable air volume (VAV) and other terminal systems maintains a static pressure at a measurement point. The most common application for static-pressure control is fan output control in VAV systems. The pressure sensor must be properly placed to maintain optimum pressure throughout the supply duct. Experience indicates that performance is satisfactory when the sensor is located at 75 to 100% of the distance from the first to the most remote terminal. If the sensor is located at less than 100% of the distance, the control set point should be adjusted higher to account for the pressure loss between the sensor and the remote terminal (Figure 5). Care must be taken in selecting the reference sensor location. Controller upset from opening and closing doors, elevator shafts, and other sources of air turbulence should also be prevented. The pressure selected provides a minimum static pressure to all air terminal units during all supply fan design conditions.

Multiple static sensors (Figure 6) are required when more than one branch duct runs from the supply fan. The sensor with the highest static requirement controls the fan. Because duct run-outs may vary, a control that uses individual set points for each measurement is preferred.

VAV systems typically incorporate a duct static-pressure control loop to control the supply fan speed output. In a single-duct VAV system, the duct static pressure set point is usually selected by the designer. The sensor should be located in the ductwork where the established set point ensures proper operation of the zone VAV boxes under varying load (supply airflow) conditions. A shortcoming of this approach is that static-pressure control is based on the

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readings of a single sensor that is assumed to represent the pressure available to all VAV boxes. If the sensor malfunctions or is placed in a location that is not representative, operating problems will result.

An alternative approach to supply fan control in a VAV system uses flow readings from the direct digital control (DDC) zone terminal boxes to integrate zone VAV requirements with supply fan operation. Englander and Norford (1992) suggest that duct static pressure and fan energy can be reduced without sacrificing occupant comfort or adequate ventilation. They compared modified PI and heuristic control algorithms using simulation and demonstrated that either static pressure or fan speed can be regulated directly using a flow error signal from one or more zones. They noted that component modeling limitations constrained their results primarily to a comparison of the control algorithms. The results show that both PI and heuristic control schemes work, but the authors suggest that a hybrid of the two might be ideal.

Supply fan warm-up control for systems having a return fan must prevent the supply fan from delivering more airflow than the return fan maximum capacity during warm-up mode (Figure 7).

Return fan static control from returns having local (zoned) flow control is identical to supply fan static control (Figure 5). Return fan control for VAV systems is required for proper building pressurization and minimum outside air. The return fan is controlled to maintain exhaust and return air plenum pressure. The exhaust air damper is controlled to maintain building static pressure (Figure 8). This ensures that the supply fan does not pull in outside air backward through the exhaust air dampers (Seem et al. 2000).

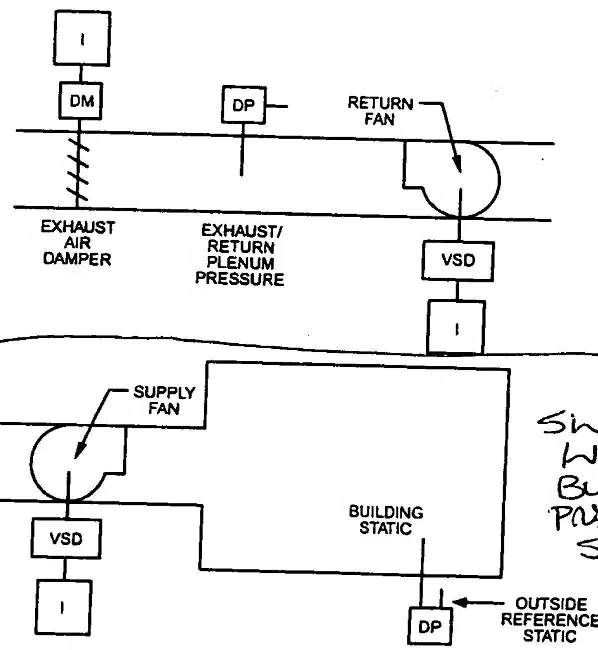
Airflow tracking uses duct airflow measurements to control the return air fans (Figure 9). Typical sensors, called flow stations, are multiple-point, pitot tube, and averaging. Provisions must be made for exhaust fan switching to maintain pressurization of the building. Warm-up is accomplished by setting the return airflow equal to the supply fan airflow, usually with exhaust fans turned off and limiting supply fan volume to return fan capability. During night cooldown, the return fan operates in the normal mode.

VAV systems that use return or relief fans require control of airflow through the return or relief air duct systems. Return fans are commonly used in VAV systems to help ensure adequate air distribution and acceptable zone pressurization. In a return fan VAV system, there is significant potential for control system instability because of the interaction of control variables (Avery 1992). In a typical system, these variables might include supply fan speed, supply duct static pressure, return fan speed, mixed air temperature, outside and return air damper flow characteristics, and wind pressure effect on the relief louver. The interaction of these variables and the selection of control schemes to minimize or eliminate interaction must be considered carefully. Mixed air damper sizing and selection are particularly important. Zone pressurization, building construction, and outside wind velocity must be considered. The resultant design helps ensure proper air distribution, especially through the return air duct. Kettler (1995) suggests that small errors in sensing total airflow and return flow can cause significant errors in control of the differential flow, making this approach unsatisfactory for minimum outside ventilation control.

Sequencing fans for VAV systems reduces airflow more than other methods and results in greater operating economy and more stable fan operation if airflow reductions are significant. Alternating fans usually provides greater reliability. Centrifugal fans are controlled to keep system disturbances to a minimum when additional fans are started. The added fan is started and slowly brought to capacity while the capacity of the operating fans is simultaneously reduced. The combined output of all fans then equals the output before fan addition.

Vaneaxial fans usually cannot be sequenced in the same manner as centrifugal fans. To avoid stall, the operating fans must be reduced to some minimum level of airflow. Then, additional fans may be started and all fans modulated in parallel to achieve equilibrium.

Unstable fan operation in VAV systems can usually be avoided by proper fan sizing. However, if airflow reduction is large (typically over 60%), fan sequencing is usually required to maintain airflow in the fan's stable range.



SINGLE INTERNAL BUILDING PRESSURE SENSOR

Fig. 8 Duct Static Control of Return Fan

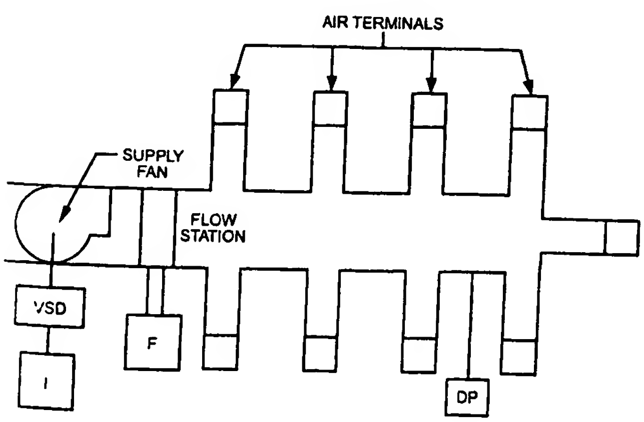


Fig. 7 Supply Fan Warm-Up Control

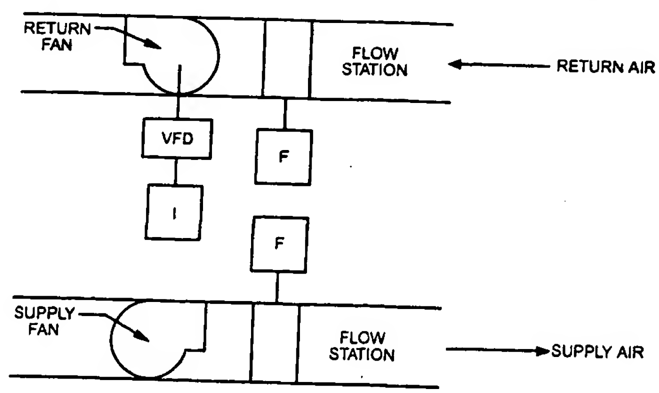


Fig. 9 Airflow Tracking Control

Humidification can be achieved by adding moisture to supply air. Evaporative pans (usually heated), steam jets, and atomizing spray tubes are all used for space humidification. A space or return air humidity sensor provides the necessary signal for the controller. A humidity sensor in the duct should be used to minimize moisture carryover or condensation in the duct (Figure 28). With proper use and control, humidifiers can achieve high space humidity, although they more often maintain design minimum humidity during the heating season.

Outside Air Control

Fixed minimum outside air control provides ventilation air, space pressurization (exfiltration), and makeup air for exhaust fans. For systems without return fans, the outside air damper is interlocked to remain open only when the supply fan operates (Figure 29). The minimum outside air damper should open quickly when the fan turns on, to prevent excessive negative duct pressurization. In some applications, the fan on-off switch opens the outside air damper before the fan is started. The rate of outside airflow is determined by the damper opening and by the pressure difference between the mixed air plenum and the outside air plenums.

For systems with return fans, two variations of fixed minimum outside air control are used. Minimum outside airflow is determined by the pressure drop across the outside damper at minimum position, and an injection fan (Figure 30) or airflow station is installed in the minimum outside air section (Figure 31). If the outside air supplied is greater than the difference between the supply and return fan airflows, a variation of economizer cycle control is used (Figure 31).

In systems using 100% outside air, all air goes to the fan and no air is returned (Figure 32). The outside air damper is interlocked and usually opens before the fan starts.

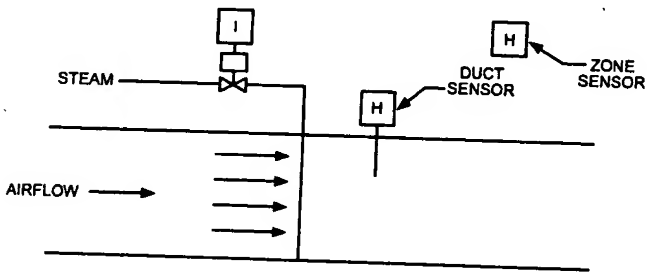


Fig. 28 Steam Jet Humidifier

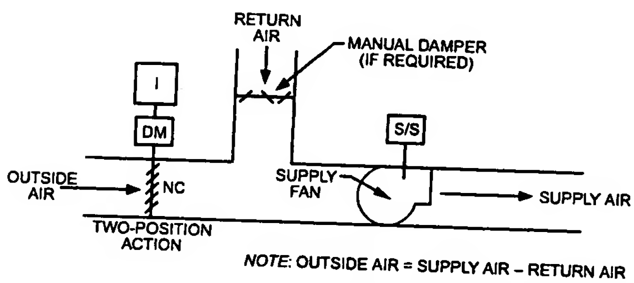


Fig. 29 Fixed Minimum Outside Air Control Without Return Fans

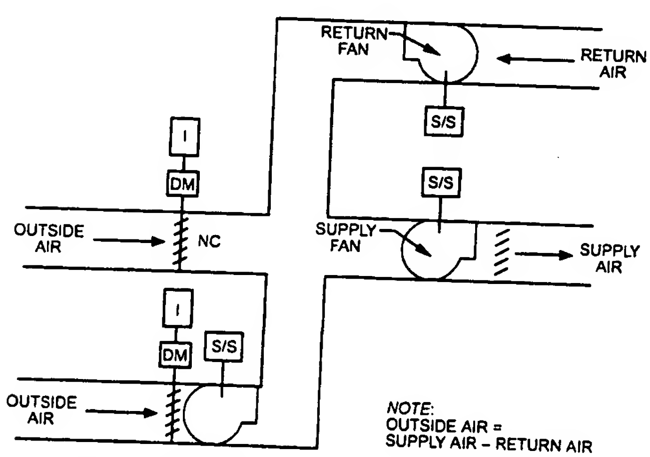


Fig. 30 Fixed Minimum Outside Air Control With Return Fans

Radiant Cooling and Heating

Radiation can be used either alone or to supplement another heater. The control strategy depends on the function performed. For a radiation-only heating application, rooms are usually controlled individually; each radiator and convector is equipped with an automatic control valve. Depending on room size, one thermostat may control one valve or several valves in unison. The thermostat can be placed in the return air to the unit or on a wall at occupant level. Return air control is generally less accurate and results in wider space-temperature fluctuations. When the space is controlled for the comfort of seated occupants, wall-mounted thermostats give the best results.

For supplemental heating applications, where perimeter radiation is used only to offset perimeter heat losses (the zone or space load is handled separately by a zone air system), outside reset of the water temperature to the radiation should be considered. Radiation can be zoned by exposure, and the compensating outside sensor can be located to sense compensated inside (outside) temperature, solar load, or both.

Radiant panels combine controlled-temperature room surfaces with central air conditioning and ventilation. The radiant panel can be in the floor, walls, or ceiling. Panel temperature is maintained by circulating water or air or by electric resistance. The central air system can be a basic one-zone, constant-temperature, constant-volume system, with the radiant panel operated by individual room control thermostats, or it can include some or all the features of

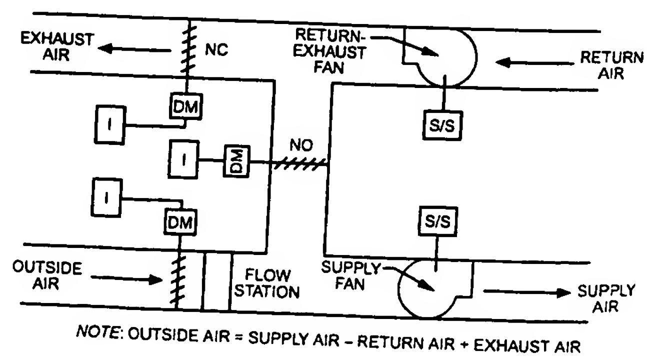


Fig. 31 Fixed Minimum Outside Air Control With Return-Exhaust Fans

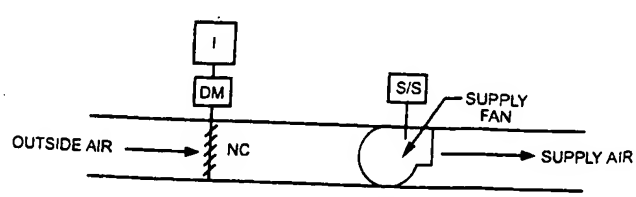


Fig. 32 100% Outside Air Control

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system provides an air change effectiveness of about 1. Therefore, the Table 6.1 values of ASHRAE *Standard* 62.1 are appropriate for design of commercial and institutional buildings when the ventilation rate procedure is used. If the indoor air quality procedure of *Standard* 62.1 is used, then actual pollutant sources and the air change effectiveness must be known for the successful design of HVAC systems that have fixed ventilation airflow rates.

ASHRAE *Standard* 129 describes a method for measuring air change effectiveness of mechanically vented spaces and buildings with limited air infiltration, exfiltration, and air leakage with surrounding indoor spaces.

DRIVING MECHANISMS FOR VENTILATION AND INFILTRATION

Natural ventilation and infiltration are driven by pressure differences across the building envelope caused by wind and air density differences due to temperature differences between indoor and outdoor air (buoyancy, or the stack effect). Mechanical air-moving systems also induce pressure differences across the envelope due to the operation of appliances, such as combustion devices, leaky forced-air thermal distribution systems, and mechanical ventilation systems. The indoor/outdoor pressure difference at a location depends on the magnitude of these driving mechanisms as well as on the characteristics of the openings in the building envelope (i.e., their locations and the relationship between pressure difference and airflow for each opening).

Stack Pressure

Stack pressure is the hydrostatic pressure caused by the weight of a column of air located inside or outside a building. It can also occur within a flow element, such as a duct or chimney, that has vertical separation between its inlet and outlet. The hydrostatic pressure in the air depends on density and the height of interest above a reference point.

Air density is a function of local barometric pressure, temperature, and humidity ratio, as described by Chapter 6. As a result, standard conditions should not be used to calculate the density. For example, a building site at 5000 ft has an air density that is about 20% less than if the building were at sea level. An air temperature increase from –20 to 70°F causes a similar air density difference. Combined, these elevation and temperature effects reduce the air density about 45%. Moisture effects on density are generally negligible, so the dry air density can be used instead, except in hot, humid climates when the air is hot and close to saturation. For example, saturated air at 105°F has a density about 5% less than that of dry air.

Assuming temperature and barometric pressure are constant over the height of interest, the stack pressure decreases linearly as the separation above the reference point increases. For a single column of air, the stack pressure can be calculated as

$$p_s = p_r - C_1 \rho g H \tag{16}$$

where

p_s = stack pressure, in. of water
p_r = stack pressure at reference height, in. of water
g = gravitational acceleration, 32.2 ft/s^2
ρ = indoor or outdoor air density, lb$_m$/ft^3
H = height above reference plane, ft
C_1 = unit conversion factor = 0.00598 (in. of water)·ft·s^2/lb$_m$

For tall buildings or when significant temperature stratification occurs indoors, Equation (16) should be modified to include the density gradient over the height of the building.

Temperature differences between indoors and outdoors cause stack pressure differences that drive airflows across the building envelope. Sherman (1991) showed that any single-zone building can be treated as an equivalent box from the point of view of stack effect, if its leaks follow the power law. The building is then characterized by an effective stack height and neutral pressure level (NPL) or leakage distribution (see the section on Neutral Pressure Level). Once calculated, these parameters can be used in physical, single-zone models to estimate infiltration.

Neglecting vertical density gradients, the stack pressure difference for a horizontal leak at any vertical location is given by

$$\Delta p_s = C_1(\rho_o - \rho_i) g (H_{NPL} - H)$$
$$= C_1 \rho_o \left(\frac{T_o - T_i}{T_i}\right) g (H_{NPL} - H) \tag{17}$$

where

T_o = outdoor temperature, °R
T_i = indoor temperature, °R
ρ_o = outdoor air density, lb/ft^3
ρ_i = indoor air density, lb/ft^3
H_{NPL} = height of neutral pressure level above reference plane without any other driving forces, ft

Chastain and Colliver (1989) showed that when there is stratification, the average of the vertical distribution of temperature differences is more appropriate to use in Equation (17) than the localized temperature difference near the opening of interest.

By convention, stack pressure differences are positive when the building is pressurized relative to outdoors, which causes flow out of the building. Therefore, in the absence of other driving forces and assuming no stack effect is within the flow elements themselves, when the indoor air is warmer than outdoors, the base of the building is depressurized and the top is pressurized relative to outdoors; when the indoor air is cooler than outdoors, the reverse is true.

In the absence of other driving forces, the location of the NPL is influenced by leakage distribution over the building exterior and by interior compartmentation. As a result, the NPL is not necessarily located at the mid-height of the building, nor is it necessarily unique. NPL location and leakage distribution are described later in the section on Combining Driving Forces.

For a penetration through the building envelope for which (1) there is a vertical separation between its inlet and outlet and (2) the air inside the flow element is not at the indoor or outdoor temperature, such as in a chimney, more complex analyses than Equation (17) are required to determine the stack effect at any location on the building envelope.

Wind Pressure

When wind impinges on a building, it creates a distribution of static pressures on the building's exterior surface that depends on the wind direction, wind speed, air density, surface orientation, and surrounding conditions. Wind pressures are generally positive with respect to the static pressure in the undisturbed airstream on the windward side of a building and negative on the leeward sides. However, pressures on these sides can be negative or positive, depending on wind angle and building shape. Static pressures over building surfaces are almost proportional to the velocity head of the undisturbed airstream. The wind pressure or velocity head is given by the Bernoulli equation, assuming no height change or pressure losses:

$$p_w = C_2 C_p \rho \frac{U^2}{2} \tag{18}$$

where

p_w = wind surface pressure relative to outdoor static pressure in undisturbed flow, in. of water
ρ = outside air density, lb$_m$/ft^3 (about 0.075)

U = wind speed, mph
C_p = wind surface pressure coefficient, dimensionless
C_2 = unit conversion factor = 0.0129 (in. of water)·ft³/lb$_m$·mph²

C_p is a function of location on the building envelope and wind direction. Chapter 16 provides additional information on the values of C_p.

Most pressure coefficient data are for winds normal to building surfaces. Unfortunately, for a real building, this fixed wind direction rarely occurs, and when the wind is not normal to the upwind wall, these pressure coefficients do not apply. A harmonic trigonometric function was developed by Walker and Wilson (1994) to interpolate between the surface average pressure coefficients on a wall that were measured with the wind normal to each of the four building surfaces. This function was developed for low-rise buildings three stories or less in height. For each wall of the building, C_p is given by

$$C_p(\phi) = \frac{1}{2}\{[C_p(1)+C_p(2)](\cos^2\phi)^{1/4} + [C_p(1)-C_p(2)](\cos\phi)^{3/4} + [C_p(3)+C_p(4)](\sin^2\phi)^2 + [C_p(3)-C_p(4)]\sin\phi\} \qquad (19)$$

where

$C_p(1)$ = pressure coefficient when wind is at 0°
$C_p(2)$ = pressure coefficient when wind is at 180°
$C_p(3)$ = pressure coefficient when wind is at 90°
$C_p(4)$ = pressure coefficient when wind is at 270°
ϕ = wind angle measured clockwise from the normal to Wall 1

The measured data used to develop the harmonic function from Akins et al. (1979) and Wiren (1985) show that typical values for the pressure coefficients are $C_p(1) = 0.6$, $C_p(2) = -0.3$, $C_p(3) = C_p(4) = -0.65$. Because of the geometry effects on flow around a building, the application of this interpolation function is limited to low-rise buildings that are of rectangular plan form (i.e., not L-shaped) with the longest wall less than three times the length of the shortest wall. For less regular buildings, simple correlations are inadequate and building-specific pressure coefficients are required. Chapter 16 discusses wind pressures for complex building shapes and for high-rise buildings in more detail.

The wind speed most commonly available for infiltration calculations is the wind speed measured at the local weather station, typically the nearest airport. This wind speed needs to be corrected for reductions caused by the difference between the height where the wind speed is measured and the height of the building and reductions due to shelter effects.

The reference wind speed used to determine pressure coefficients is usually the wind speed at the eaves height for low-rise buildings and the building height for high-rise buildings. However, meteorological wind speed measurements are made at a different height (typically 33 ft) and at a different location. The difference in terrain between the measurement station and the building under study must also be accounted for. Chapter 16 shows how to calculate the effective wind speed U_H from the reference wind speed U_{met} using boundary layer theory and estimates of terrain effects.

In addition to the reduction in wind pressures due to the reduction in wind speed, the effects of local shelter also act to reduce wind pressures. The shielding effects of trees, shrubbery, and other buildings within several building heights of a particular building produce large-scale turbulence eddies that not only reduce effective wind speed but also alter wind direction. Thus, meteorological wind speed data must be reduced carefully when applied to low buildings.

Ventilation rates measured by Wilson and Walker (1991) for a row of houses showed reductions in ventilation rates of up to a factor of three when the wind changed direction from perpendicular to parallel to the row. They recommended estimating wind shelter for winds perpendicular to each side of the building and then using the interpolation function in Equation (20) to find the wind shelter for intermediate wind angles:

$$s = \frac{1}{2}\left\{\begin{matrix}[s(1)+s(2)]\cos^2\phi + [s(1)-s(2)]\cos\phi \\ + [s(3)+s(4)]\sin^2\phi + [s(3)-s(4)]\sin\phi\end{matrix}\right\} \qquad (20)$$

where

s = shelter factor for the particular wind direction ϕ
$s(i)$ = shelter factor when wind is normal to Wall i (i = 1 to 4, for four sides of a building)

Although the above method gives a realistic variation of wind shelter effects with wind direction, estimates for the numerical values of wind shelter factor s for each of the four cardinal directions must be provided. Table 8 in the section on Residential Calculation Examples lists typical shelter factors. The wind speed used in Equation (18) is then given by

$$U = sU_H \qquad (21)$$

The magnitude of the pressure differences found on the surfaces of buildings varies rapidly with time because of turbulent fluctuations in the wind (Etheridge and Nolan 1979; Grimsrud et al. 1979). However, the use of average wind pressures to calculate pressure differences is usually sufficient to calculate average infiltration values.

Mechanical Systems

The operation of mechanical equipment, such as supply or exhaust systems and vented combustion devices, affects pressure differences across the building envelope. The interior static pressure adjusts such that the sum of all airflows through the openings in the building envelope plus equipment-induced airflows balance to zero. To predict these changes in pressure differences and airflow rates caused by mechanical equipment, the location of each opening in the envelope and the relationship between pressure difference and airflow rate for each opening must be known. The interaction between mechanical ventilation system operation and envelope airtightness has been discussed for low-rise buildings (Nylund 1980) and for office buildings (Persily and Grot 1985a; Tamura and Wilson 1966, 1967b).

Air exhausted from a building by a whole-building exhaust system must be balanced by increasing the airflow into the building through other openings. As a result, the airflow at some locations changes from outflow to inflow. For supply fans, the situation is reversed and envelope inflows become outflows. Thus, the effects a mechanical system has on a building must be considered. Depressurization caused by an improperly designed exhaust system can increase the rate of radon entry into a building and interfere with the proper operation of combustion device venting or other exhaust systems. Depressurization can also force moist outdoor air through the building envelope; for example, during the cooling season in hot humid climates, moisture may condense within the building envelope. A similar phenomenon, but in reverse, can occur during the heating season in cold climates if the building is depressurized.

The interaction between mechanical systems and the building envelope also pertains to systems serving zones of buildings. The performance of zone-specific exhaust or pressurization systems is affected by the leakage in zone partitions as well as in exterior walls.

Mechanical systems can also create infiltration-driving forces in single-zone buildings. Specifically, some single-family houses with central forced-air duct systems have multiple supply registers, yet only a central return register. When internal doors are closed in these houses, large positive indoor-outdoor pressure differentials are created for rooms with only supply registers, whereas the room with the return duct tends to depressurize relative to outside. This is caused by the resistance of internal door undercuts to flow from the

Y

supply register to the return (Modera et al. 1991). The magnitudes of the indoor/outdoor pressure differentials created have been measured to average 0.012 to 0.024 in. of water (Modera et al. 1991).

Building envelope airtightness and interzonal airflow resistance can also affect the performance of mechanical systems. The actual airflow rate delivered by these systems, particularly ventilation systems, depends on the pressure they work against. This effect is the same as the interaction of a fan with its associated ductwork, which is discussed in Chapter 35 of this volume and Chapter 18 of the 2004 *ASHRAE Handbook—HVAC Systems and Equipment*. The building envelope and its leakage must be considered part of the ductwork in determining the pressure drop of the system.

Duct leakage can cause similar problems. Supply leaks to the outside will tend to depressurize the building; return leaks to the outside will tend to pressurize it.

Combining Driving Forces

The pressure differences due to wind pressure, stack pressure, and mechanical systems are considered in combination by adding them together and then determining the airflow rate through each opening due to this total pressure difference. The air flows must be determined in this manner, as opposed to adding the airflow rates due to the separate driving forces, because the airflow rate through each opening is not linearly related to pressure difference.

For uniform indoor air temperatures, the total pressure difference across each leak can be written in terms of a reference wind parameter P_U and stack effect parameter P_T common to all leaks:

$$P_U = \rho_o \frac{U_H^2}{2} \quad (22)$$

$$P_T = g\rho_o\left(\frac{T_o - T_i}{T_i}\right) \quad (23)$$

where T = air temperature, °R.

The pressure difference across each leak (with positive pressures for flow into the building) is then given by

$$\Delta p = C_2 s^2 C_p P_U + HP_T + \Delta p_I \quad (24)$$

where Δp_I = pressure that acts to balance inflows and outflows (including mechanical system flows). Equation (24) can then be applied to every leak for the building with the appropriate values of C_p, s, and H. Thus, each leak is defined by its pressure coefficient, shelter, and height. Where indoor pressures are not uniform, more complex analyses are required.

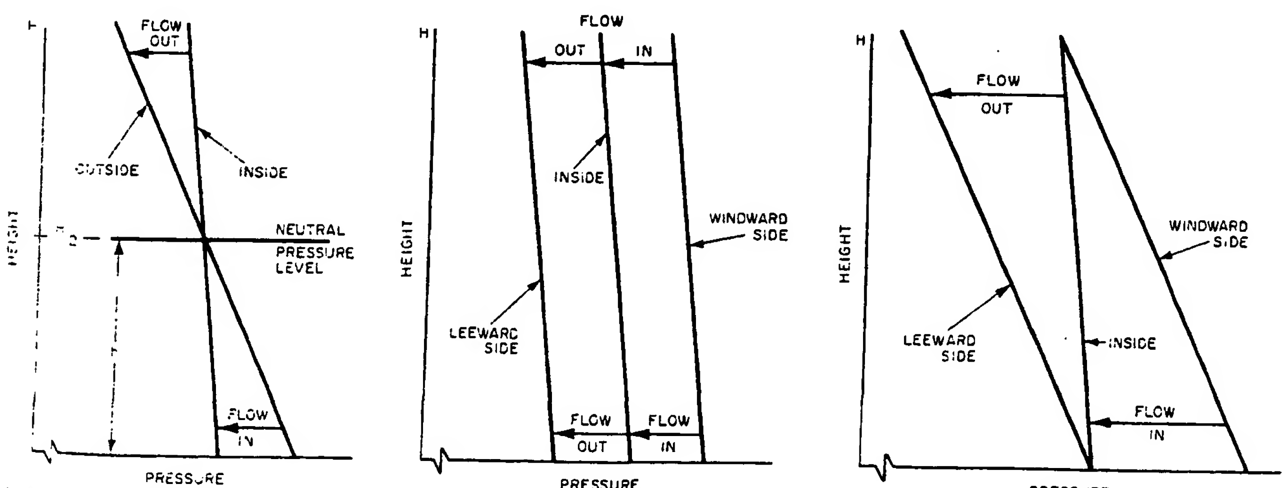
Neutral Pressure Level

The neutral pressure level (NPL) is that location or locations in the building envelope where there is no pressure difference. Internal partitions, stairwells, elevator shafts, utility ducts, chimneys, vents, operable windows, and mechanical supply and exhaust systems complicate the analysis of NPL location. An opening with a large area relative to the total building leakage causes the NPL to shift toward the location of the opening. In particular, chimneys and openings at or above roof height raise the NPL in small buildings. Exhaust systems increase the height of the NPL; outdoor air supply systems lower it.

Figure 5 qualitatively shows the addition of driving forces for a building with uniform openings above and below mid-height and without significant internal resistance to airflow. The slopes of the pressure lines are a function of the densities of the indoor and outdoor air. In Figure 5A, with inside air warmer than outside and pressure differences caused solely by thermal forces, the NPL is at mid-height, with inflow through lower openings and outflow through higher openings. Direction of flow is always from the higher to the lower pressure region.

Figure 5B presents qualitative uniform pressure differences caused by wind alone, with opposing effects on the windward and leeward sides. When the temperature difference and wind effects both exist, the pressures due to each are added together to determine the total pressure difference across the building envelope. In Figure 5B, there is no NPL because no locations on the building envelope have zero pressure difference. Figure 5C shows the combination, where the wind force of Figure 5B has just balanced the thermal force of Figure 5A, causing no pressure difference at the top windward or bottom leeward side.

The relative importance of the wind and stack pressures in a building depends on building height, internal resistance to vertical airflow, location and flow resistance characteristics of envelope openings, local terrain, and the immediate shielding of the building. The taller the building is and the smaller its internal resistance to airflow, the stronger the stack effect. The more exposed a building is, the more susceptible it is to wind. For any building, there are ranges of wind speed and temperature difference for which the building's



A. STACK ACTION ONLY WITH NEUTRAL PRESSURE LEVEL AT MID-HEIGHT

B. WIND ACTION ONLY WITH PRESSURES OF EQUAL MAGNITUDE ON WINDWARD AND LEEWARD SIDES

C. WIND AND STACK ACTION COMBINED

Fig. 5 Distribution of Inside and Outside Pressures over Height of Building

Z

infiltration is dominated by the stack effect, the wind, or the driving pressures of both (Sinden 1978a). These building and terrain factors determine, for specific values of temperature difference and wind speed, in which regime the building's infiltration lies.

The effect of mechanical ventilation on envelope pressure differences is more complex and depends on both the direction of the ventilation flow (exhaust or supply) and the differences in these ventilation flows among the zones of the building. If mechanically supplied outdoor air is provided uniformly to each story, the change in the exterior wall pressure difference pattern is uniform. With a nonuniform supply of outdoor air (for example, to one story only), the extent of pressurization varies from story to story and depends on the internal airflow resistance. Pressurizing all levels uniformly has little effect on the pressure differences across floors and vertical shaft enclosures, but pressurizing individual stories increases the pressure drop across these internal separations. Pressurizing the ground level is often used in tall buildings to reduce stack pressures across entries.

Available data on the NPL in various kinds of buildings are limited. The NPL in tall buildings varies from 0.3 to 0.7 of total building height (Tamura and Wilson 1966, 1967a). For houses, especially houses with chimneys, the NPL is usually above mid-height. Operating a combustion heat source with a flue raises the NPL further, sometimes above the ceiling (Shaw and Brown 1982).

Thermal Draft Coefficient

Compartmentation of a building also affects the NPL location. Equation (17) provides a maximum stack pressure difference, given no internal airflow resistance. The sum of the pressure differences across the exterior wall at the bottom and at the top of the building, as calculated by these equations, equals the total theoretical draft for the building. The sum of the actual top and bottom pressure differences, divided by the total theoretical draft pressure difference, equals the **thermal draft coefficient**. The value of the thermal draft coefficient depends on the airflow resistance of the exterior walls relative to the airflow resistance between floors. For a building without internal partitions, the total theoretical draft is achieved across the exterior walls (Figure 6A), and the thermal draft coefficient equals 1. In a building with airtight separations at each floor, each story acts independently, its own stack effect being unaffected by that of any other floor (Figure 6B). The theoretical draft is minimized in this case, and each story has an NPL.

Real multistory buildings are neither open inside (Figure 6A), nor airtight between stories (Figure 6B). Vertical air passages, stairwells, elevators, and other service shafts allow airflow between floors. Figure 6C represents a heated building with uniform openings in the exterior wall, through each floor, and into the vertical shaft at each story. Between floors, the slope of the line representing the inside pressure is the same as that shown in Figure 6A, and the discontinuity at each floor (Figure 6B) represents the pressure difference across it. Some of the pressure difference maintains flow through openings in the floors and vertical shafts. As a result, the pressure difference across the exterior wall at any level is less than it would be with no internal flow resistance.

Maintaining airtightness between floors and from floors to vertical shafts is a means of controlling indoor-outdoor pressure differences due to the stack effect and therefore infiltration. Good separation is also conducive to the proper operation of mechanical ventilation and smoke management systems. However, care is needed to avoid pressure differences that could prevent door opening in an emergency. Tamura and Wilson (1967b) showed that when vertical shaft leakage is at least two times the envelope leakage, the thermal draft coefficient is almost one and the effect of compartmentation is negligible. Measurements of pressure differences in three tall office buildings by Tamura and Wilson (1967a) indicated that the thermal draft coefficient ranged from 0.8 to 0.9 with the ventilation systems off.

COMMERCIAL AND INSTITUTIONAL VENTILATION REQUIREMENTS

ASHRAE *Standard* 62.1 contains requirements on ventilation and indoor air quality for commercial, institutional, and high-rise residential buildings. These requirements address system and equipment issues, design ventilation rates, commissioning and systems start-up, and operations and maintenance. The design requirements, include two alternative procedures:

- The prescriptive **ventilation rate procedure** contains a table of outdoor air ventilation requirements for a variety of space types with adjustments for air distribution in rooms and systems serving multiple spaces. These minimum outside air ventilation rates are based, in part, on research by Berg-Munch et al. (1986), Cain et al. (1983), Iwashita et al. (1989), and Yaglou et al. (1936) as well as years of experience of designers and building operators.
- The **indoor air quality procedure**, which achieves acceptable indoor air quality through the control of indoor contaminant concentrations. Such control can be realized through source control,

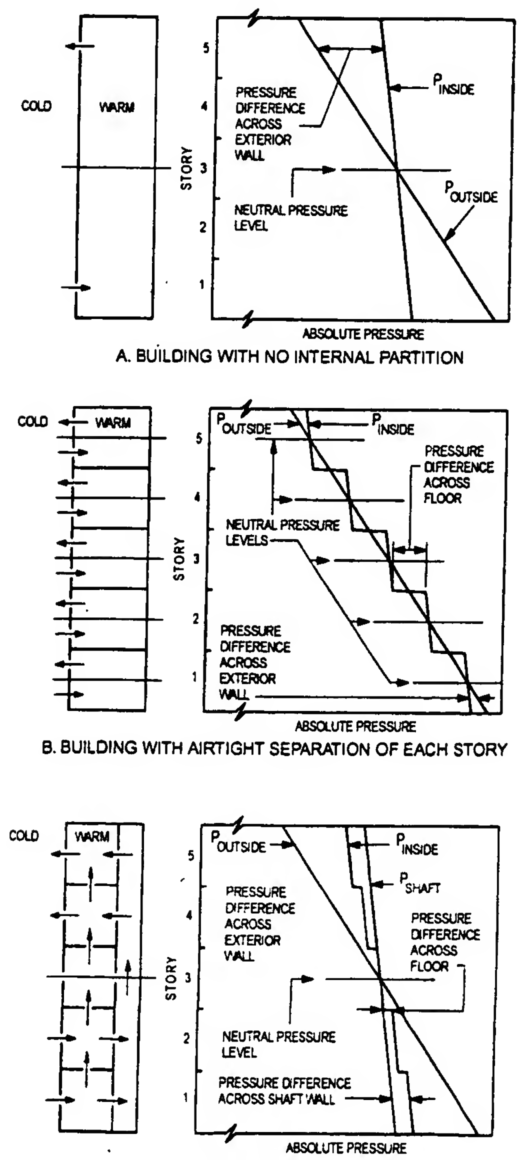


Fig. 6 Compartmentation Effect in Buildings

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